An

Investigation

of the

Critical Bearing Pressures Causing Rupture

in

Lubricating Oil Films

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Introduction.

In selecting a problem for engineering research, no field seemed so fraught with possibilities as that of lubrication. This is apparent when it is realized that no researches of a really fundamental character, with the notable exceptions of those of Kingsbury and Michell, have been undertaken since 1883. This is certainly surprising in view of the fact that the subject is of such fundamental importance in engineering practice.

The introduction of steam turbines, internal combustion engines and high speed machinery into modern industry necessitated new and more efficient means of lubrication. Activity along this line has been notable in the last few years and some really important advances have been made, especially in the investigation of that somewhat obscure quality of lubricating oils, which for want of a better name has been termed "oiliness", and also of the treatment of oils with fatty acids and colloidal graphite to increase their lubricating quality.

The present accepted theory of lubrication is based on the classic researches of Mr. Beauchamp Tower, whose first report was made to the Institute of Mechanical Engineers in 1883 and his second in 1884. Mr. Tower's experiments were conducted with great care and under a great variety of conditions. The results obtained failed to agree with the results of previous investigations and this lack of concordance led Mr. Tower to conclude that it was due to the character of the supply of the lubricant. It also fortunately caused him to investigate the phenomena of flooded lubrication and his researches in this important phase brought to light a fundamental fact that had not been clearly recognized up to that time, namely: That with flooded lubrication, the bearing surfaces are separated by an oil film of definite thickness.

Using the data of Mr. Tower's work, Mr. Osborne Reynolds developed the hydrodynamical theory of lubrication and published it in the Phil. Trans. of the Royal Society in 1886. It seems, however, that Prof. Stokes and Lord Rayleigh simultaneously propounded a similar theory.

It is curious to note that N. Petroff, a Russian experimenter, published his "Neue Theorie der Reibung", which is frankly an exposition of the hydrodynamical theory, in Russian

in 1883 and in German in 1887. Yet no note was taken of his work by English writers until quite recently when a translation was published in The Engineer in 1918. German writers on the other hand refer to his work frequently and Prof. A. Sommerfeld makes this terse remark in regard to Petroff's formula for the moment of friction: "Dies ist die Formel Petroffs, des Vaters der hydrodynamischen Schmiermitteltheorie".

The literature on the subject of lubrication is very extensive. Although numerous treatises and articles have been published there does not seem to be a common agreement on the more involved points in the mechanism of lubrication. That this disagreement is a matter of moment, is evinced by the statement to the writer, by a research engineer of international reputation, that in the practical designing of bearings the practice departs materially from the theory and the design proceeds entirely along empirical lines. He further pointed out that the laws of film lubrication are very imperfectly established and suggested that if these obscure points were cleared up the whole subject could be treated in a more rational manner.

Although there are certain aspects of the problem that are involved and obscure there are certain fundamental notions which are accepted as correct and which can be used with assurance.

Fundamental Conceptions.

It is the function of the lubricant to insinuate itself between the sliding members of a machine and by so doing reduce the friction between the parts. It often happens, however, that the lubricant fails entirely to reach the parts that it is intended to lubricate, or only partly performs its function due to inadequate supply, to intermittent or to improper application.

The field of application, for purposes of discussion, may be divided into three classes in accordance with the degree of lubrication attained in each, namely:

- I Unlubricated surfaces. (Dry friction).
- II Partially lubricated surfaces. (Oily friction).
- III Completely lubricated surfaces. (Viscous friction).

Dry Friction.

The so-called laws of dry friction were first enunciated by C. A. Coulomb, in 1781, and confirmed by A. J. Morin, in 1834, and have been found to hold good within very wide limits. According to these laws the assumption is made that the friction is:

- I Proportional to the normal pressure between the surfaces in contact.
- II Independent of the area of these surfaces.

III Independent of the velocity with which the surfaces slide upon one another.

IV Dependent on the nature of the surfaces in contact.

These statements do in fact sensibly represent the results of experiments for pressures and velocities most commonly occuring. If these assumptions are correct, dry friction may be measured simply in terms of total pressure between the surfaces, multiplied by the coefficient depending on the nature of the surfaces in contact.

Coulomb recognized the fact that as greater effort was necessary to set a body in motion than to keep it in motion, so we must distinguish between static friction (the friction of rest) and kinetic friction (the friction of motion). He also pointed out that the value of static friction often depended not only on the intensity of pressure, but also upon the time the body had been at rest. At higher velocities than are commonly encountered in engineering practice, it has been shown by several experimenters that the friction is much less than at lower velocities. This fact can be explained by the possibility of the formation of an air film at these high velocities which acting as a lubricant materially reduces the friction. No increase in the friction at slow speed was detected until the experiments of Prof. Fleming Wilson (Phil. Trans. 1877) demonstrated conclusively that at extremely low velocities, of the order of 0.0002 feet per second, there is a sensible increase of frictional resistance above that at ordinary velocities.

These experiments point to the conclusion that the coefficient of kinetic friction gradually increases as the velocity grows smaller and passes without discontinuity into that of static friction.

The Coefficient of Friction.

The coefficient of friction is defined as that value which, when multiplied by the pressure normal to the surface in contact, gives the maximum frictional resistance to motion:

Let $F =$ Total frictional resistance
$P =$ Normal pressure on surfaces in contact
$f =$ Coefficient of friction

Then $$f = \frac{F}{P} \tag{1}$$

Although the term coefficient of friction has no physical meaning, it supplies a convenient and conventional method of discussing results in which friction is involved.

Lord Rayleigh (Phil. Mag. Vol. 35, Feb. 1918) and Prof. W. B. Hardy (Phil. Mag. Vol. 38, July 1919) have demonstrated the existence of an invisible film that is present on apparently clean surfaces. An interesting and simple experiment performed

by means of an "optical flat" and a Swedish gage, will disclose the fact that such an invisible film exists even though the glass has been carefully cleaned. If the gage is placed on the glass and the glass inclined it will slide with great freedom but if any pressure is applied, particularly if the gage is rotated at the same time, the gage will adhere with considerable force. If the experiment is repeated with the glass inverted and the gage held against the under side of the glass, this film makes itself manifest by the presence of light bands, as the optical flat is in substance an interferometer. The width and curvature of these bands are a measure of the thickness of the film. With slight pressure the gage can be slid along the glass freely and the light bands are uniform in width and curvature, but when pressure is applied they are seen to widen and bow to greater extent and when the critical pressure is reached, i.e., one that is sufficient to rupture and dislodge the film, the disc will adhere and the bands disappear.

The cause of the inconsistency of published data is due, no doubt, to the presence of these contaminating films on apparently clean surfaces, and to molecular adhesion which is a somewhat obscure and unmeasurable factor.

Oily Friction.

The stage of lubrication in which an extremely thin film exists or in which there is perhaps a viscous film formed when the lubricant is applied, followed by a period in which there is metallic contact, is known as the intermediate stage and it is in this stage that the "oiliness" of the lubricant plays the leading role. The real nature of this kind of friction is a problem in molecular physics and physical chemistry and is so involved that there is, at the present time, no adequate explanation of its vagaries but an examination of the trend of thought on this important phase is certainly of interest.

Nature of Oiliness.

This matter of "oiliness" has recently been the subject of much study, experimentation and conjecture. There is much divergence of opinion as to its real nature. The outstanding fact is that some lubricants have more effect in reducing friction between surfaces intimately in contact than others. Prof. Hardy (Phil. Mag. Vol. 38, July 1919) states that with a true lubricant, a film of the order of one millionth of a millimeter in thickness is sufficient to cause two glass plates to slide over one another with the minimum of friction, and concludes that lubrication depends wholly upon the chemical constitution of the fluid and that a true lubricant is always a fluid which is adsorbed by the solid face.

Lubricants are colloids and it is well known that whenever a colloid comes in contact with a surface it fails to maintain a uniform special distribution. There is a greater concentration at the point of contact than that which obtains in the

interior of the colloidal mass. This property of concentration on a surface is known as "adsorption".

Mr. Deeley (Oil Engineering and Finance, 14 Jan. 1922, p. 64) holds the view that "Oiliness would appear to be an effect produced by the lubricant upon the metallic surfaces with which it is in contact, rather than a property dependent upon any particular physical property of the lubricant. It would appear that the unsaturated molecules of the lubricant enter into firm physico-chemical union with the metallic surfaces, thus forming a friction surface, which is a compound of oil and metal. This solid surface would also appear in the case of metallic surfaces to be much more than one molecule thick, the oil penetrating some little distance into the metal, and altering its physical properties or, as a result of abrasion, forming a paste of metal plus oil between surfaces covered by oil layers one molecule thick."

Other authorities disagree with this entirely and maintain that there is always a film of the lubricant itself between the two surfaces. The concensus of opinion seems to be that oiliness may be defined in terms of the chemical constituents of the lubricant and of the metal to which it is applied, but in the light of present knowledge this cannot be stated in exact terms.

As a matter of conjecture it would appear that photomicrography might clear up the metallic paste theory and perhaps experiments with bearing materials which are impervious to chemical action, such as platinum or zirconium, would throw further light on the subject. There can be no doubt that oil under pressure penetrates into the bearing metal to a considerable depth and that it remains so absorbed for a period, gradually reappearing on the surface. Dr. Hele-Shaw (Oil Engineering and Finance, 14 Jan. 1922) records an instance where some specimens were prepared for photomicrographs and it was found that a certain phosphor-bronze specimen would not retain its polish but would shortly become dull. On investigation it was discovered that this particular specimen was from a slipper which had been in service for some years under great pressure on a lubricated surface, and the recurring dullness was due to the gradual reappearance of the lubricant on the surface. The new theory of the atom might possibly offer an explanation of the real factor causing "oiliness".

The Production of Oily Mixtures.

Oils of vegetable and animal origin of the same viscosity are superior to straight mineral oils in lubricating qualities. The oil trade has long known how to produce oily mixtures. The usual method of producing oils in which a high degree of oiliness is regarded as essential is by simply adding a certain amount of fixed oil to the mineral oil. For instance, in marine steam engine bearings of the open type, it has been common practice to use from 10 to 25 percent thickened or blown rape oil mixed with

the mineral oil.

Wells and Southcombe (Trans. Soc. Chem. Industry, Vol. [illegible], 1[illegible] Mar. 19[illegible]) have put forward the theory that the superior lubricating quality in these compounded oils is due to the presence of free fatty acids in the added fixed oils and that the same result could be produced by simply adding free fatty acids as such. They have developed a process of treating mineral oils to which they have given the name of "Germ Process". This process appears to consist essentially of adding known amounts of free fatty acids to the mineral oil in the form of what they term "Oil Essence".

Other investigators have confirmed their contentions experimentally and it is pretty well established that the judicious admixture of free fatty acids to straight mineral oils will increase the lubricating quality of the latter.

The presence of acids of any sort in lubricating oils has been regarded as highly undesirable on account of the possibility of it causing corrosion. It is certainly less objectionable, however, to add known and controllable amounts of fatty acids that will remain inert, than to use fixed oils containing unknown quantities of fatty glycerides which are potentially capable of generating, by hydrolysis, large amounts of organic acids.

<u>The Nature of Viscosity.</u>

In the mathematical development of the fluid friction, viscosity is properly regarded as the controlling variable and therefore its nature must be understood before proceeding with the theory.

Everyone understands in a general way what viscosity is, but even a casual reading of the literature on the subject brings to light the fact that there is a great deal of misconception not only as to the true nature of viscosity but also as to the part it plays in lubrication.

Viscosity is, by definition, the state or property of being viscous, that is, that property in a liquid that resists flow. Thus pitch and molasses are highly viscous, while water and alcohol are much less so. In a physical sense this difference of fluidity is regarded as internal friction between the infinitely minute layers of the liquid.

Some writers regard viscosity as molecular cohesion but this view does not seem to be justified as it has been found by experiment that the molecules of water move about one another with great freedom, and water has a very low viscosity in consequence, but, according to the calculation of Young and Dupré, a force of about 25,000 atmospheres is required to tear the molecules apart. There does not therefore seem to be any relation between viscosity and molecular cohesion.

The measure of resistance to distortional motion in the liquid is called the coefficient of viscosity and can be found thus:

$$R = \frac{\mu \; v}{\Delta} \qquad (2)$$

and

$$\mu = \frac{R \Delta}{v} \qquad (3)$$

when
μ = Coefficient of viscosity.
Δ = Distance between layers.
v = Velocity of shear at distance Δ .
R = Shearing force over unit area between two layers.

The condition of viscous flow between two parallel plane surfaces is of prime importance when the subject of lubrication is considered and such a premise lends itself readily to mathematical treatment, yet such conditions are not those under which the viscosity can be most easily and accurately determined. The most concordant values have been obtained by measuring the rate of flow through capillary tubes.

Carefully conducted experiments by Jean Louis Marie Poiseuille a celebrated French physiologist who lived from 1797 to 1869, demonstrated that the volume of liquid passed by a capillary tube is directly proportional to the pressure urging it along and to the fourth power of the radius of the tube and inversely to its length. This is known as Poiseuille's law. In recognition of Poiseuille's work which was fundamental in character and on account of his many other contributions to science, it has been recommended that the practical unit of absolute viscosity be called the POISE. This unit is defined as THE FORCE THAT WILL MOVE A UNIT AREA OF PLANE SURFACE AT UNIT SPEED RELATIVE TO ANOTHER PARALLEL SURFACE FROM WHICH IT IS SEPARATED BY A LAYER OF THE LIQUID OF UNIT THICKNESS. The values used in defining this unit are expressed in the centimeter-grams-seconds system, and it so happens that the viscosity of water at 68.4° F. (20.2° C.) is exactly one one-hundredth of a poise. This subdivision has been named the centi-poise after the manner of the decimal system.

The mathematical treatment of viscosity has been simplified by assuming that the boundaries do not slip and making the coefficient of adhesion equal to infinity. This assumption seems justified since Warburg (Annalen der Physik und Chemie, Vol. 140, p. 367) in a series of experiments conducted with capillary tubes, showed that there was no slip at the bounding surfaces. Although Warburg demonstrated that this was true under the conditions of his test, there was no evidence that the adhesion was sufficient to prevent slippage if the shearing force were sufficiently increased.

A committee of the American Society of Mechanical Engineers (Jour. A. S. M. E., Vol. 41, p. 537) attempted to cause slippage with films as thin as 0.00025 in. and rates of shear up to 260,000 radians per second, at atmospheric temperature and

pressure, but could find no indication of slippage or deviation from the ordinary law of viscous friction that could not be attributed to inaccuracy of the fitted surfaces.

Oils grow less viscous with increase of temperature, but the exact law of variation is unknown. The erroneous assumption is often made that the viscosities determined by the efflux instruments are directly proportional to the absolute viscosities, and also that the viscosities of all oils are the same at high temperature. It is true that they approach each other at high temperatures but they are far from being the same. Temperature is the controlling variable causing changes of viscosity but there is another factor often lost sight of, that is, the change of viscosity and density with pressure. Although it has long been known that there was such a change, it has only recently been determined quantitatively. (Report of the Lubricants and Lubrication Inquiry Committee, British Department of Industrial Research).

When the pressures are moderate, this change of viscosity with pressure can be neglected but when they are high it certainly should be taken into consideration.

The increase of apparent viscosity for an increase of pressure from zero to six or seven tons per square inch was found to be from three to four hundred percent greater in the fixed oils and from twelve to thirty hundred percent greater in the mineral oils.

Unfortunately the capillary tube method of determining viscosities is hardly suitable for commercial tests and neither are the other types of viscosimeters commonly employed in the physical laboratory. The commercial testing of lubricants for viscosity is practically limited to the Saybolt, Redwood, Engler and Barbey viscosimeters. The Saybolt viscosimeter is largely used in America, while the Redwood and Engler instruments are commonly employed in England and Germany respectively and the Barbey in France. All of these instruments are of the efflux type and the viscosity is an arbitrary unit depending on the time of flow of a measured quantity of oil at a reference temperature through a short tube of small diameter.

It is very evident that the use of so many arbitrary units leads to confusion. It is necessary in scientific work to reduce these arbitrary units to absolute viscosities. This can be done for the Saybolt, Engler and Redwood viscosimeters by equation No. [illegible], derived by Dr. Herschel of the Bureau of Standards (Bureau of Standards Technologic Paper, No. 11[illegible]).

$$\frac{\mu}{\gamma} = A\,S - \frac{B}{S} \qquad (4)$$

where μ = absolute viscosity in C. G. S. units.
γ = density in grams per cu. cm.
S = efflux time in seconds.
A and B = instrumental constants.

Viscosimeter	Instrumental Constants. A	B
Saybolt	0.00220	1.80
Engler	0.00147	3.74
Redwood (No. 1)	0.00260	1.[illegible]
Redwood (No. 2)	0.02390	1.[illegible]4

When the Saybolt Universal Viscosimeter of standard dimensions is used the equation (4) reduces to the form:

$$\mu = \left[0.0022\,S - \frac{1.8}{S}\right]\gamma \qquad (5)$$

This equation can be used with confidence provided the dimensions of the instrument used are checked against the standard. It should be noted, however, that the orifice tube can very easily be injured by the thermometer used in stirring or by corrosion, thus seriously disturbing the calibration.

The densities at any temperature can be calculated by an equation of the form:

$$\gamma = \gamma' - k\,t' \qquad (6)$$

where γ = density at any temperature.
γ' = density at 60° F.
k = change of density per degrees F.
t' = temperature above 60° F.

It should be noted that the coefficient k changes with material changes in the gravity. It has the value 0.[illegible]0767 for the range of gravities used in the experiments of the author.

The density required can be determined by the pyknometer (specific gravity bottle) or more simply by a hydrometer. If the Baumé hydrometer is used the reading can be reduced to specific gravity by the use of the tables published in Circular No. 57 of the Bureau of Standards.

Viscous Friction.

When a bearing is properly designed, the supply of lubricant copious and the speed high enough to form a film, the resistance encountered is entirely due to the viscosity of the fluid.

The nature and law of fluid friction has been extensively treated by mathematicians, physicists and engineers. The hydrodynamic treatment by Osborne Reynolds has been extended and improved by Lord Rayleigh (Phil. Mag. Vol. [illegible], Jan. [illegible]), H. M. Martin (Engineering, Vol. [illegible] and [illegible]), Sommerfeld (Zeit. für Math. u. Physik, Vol. 50, 1904) and others.

A study of the theoretical and experimental considerations in a journal operating under conditions of flooded lubrication leads to the conclusion that the resistance encountered is:

I Proportional to the viscosity of the lubricant.
II Proportional to the speed.
III Proportional to the area.
IV Independent of the material of the bearing surfaces.
V Nearly independent of the load.

When we consider the phenomena occurring when a journal rotates in a bearing with a film of lubricant interposed between the two bearing surfaces there is a resistance to relative motion of these surfaces due to the shearing or transverse distortion of the oil film.

Referring to Fig. I, let q be the shear stress and $\frac{S}{\Delta}$ be the shear strain. We can then write $q = G\frac{S}{\Delta}$, where G is the modulus of transverse elasticity.

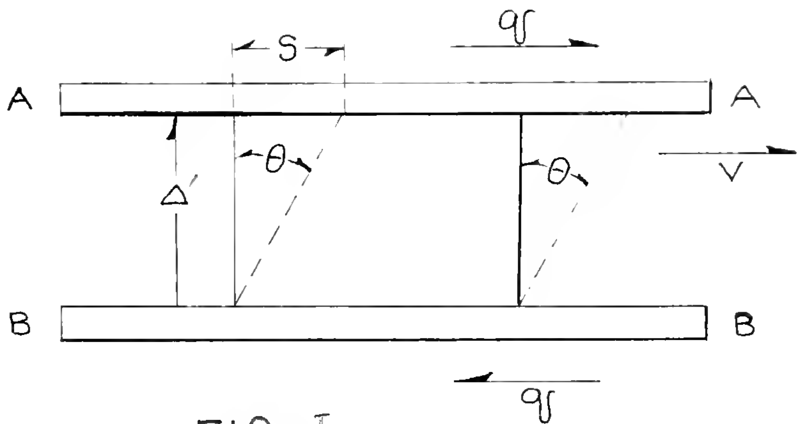


FIG. I

If the surfaces AA and BB are separated by a film having a thickness Δ, and AA moves with a speed V relative to BB, the resistance depends on the magnitude of the rate of increase of the angle θ with time.

If a is the area of either liquid surface, the force required to maintain a steady speed V is:

$$F = q\,a = \frac{\mu\, a\, V}{\Delta'} \qquad (7)$$

where μ is the coefficient of viscosity of the liquid.

A film of uniform thickness Δ', surrounding a journal of

a diameter d and a length ℓ, assuming that the coefficient of adhesion is infinite, will oppose a resistance to the rotation of the former having a peripheral speed V in accordance with the following formula:

$$F = \frac{\mu \pi d \ell}{\Delta'} \qquad (a)$$

In such a case the friction would not depend on the load, neither does it depend on the fluid pressure to which it is momentarily subjected, but depends only on the area of the fluid to be sheared and its viscosity, but it should be remembered that the viscosity changes not only with the temperature but also with the pressure.

It is apparent from the form of the equation that as Δ' grows smaller, F grows larger in the case where the film Δ' surrounding the journal is of a uniform thickness. Unfortunately there are no cases in practice when this is strictly true. In steam turbine practice, where high speed is encountered, this condition is approached but probably never quite reached.

The eccentricity of the journal in the bearing varies with the speed. It is large at low speeds and decreases as the speed increases. At the lower speed the decrease in frictional resistance which would be expected from the increase of film thickness on one side is more than offset by the thinning of the film on the other side. The friction on the whole, therefore, grows greater as the journal becomes more eccentric.

If we consider the cycle of events in a bearing arranged for flooded lubrication, it is evident that when the machine is at rest there will be metal to metal contact between the journal and the bearing and this contact will continue until sufficient speed has been attained to form the film. When this film has been formed the eccentricity will be proportionally great but will diminish as the speed increases. It follows then, that the frictional resistance will be larger at the lower speed due to this eccentricity. As the speed increases and the eccentricity grows less the friction should increase with the speed but should decrease as the journal approaches concentricity. When the condition of concentricity is approached the change in the friction, due to change in eccentricity, will be inappreciable. In this condition of operation the friction would increase substantially in proportion to the velocity of rubbing, were it not for the fact that the increase of temperature causes a decrease in the viscosity and, therefore, the friction increases less rapidly than in exact proportion to the speed.

At very low speed the coefficient of friction is proportional to the square root of the speed but this relationship changes as the speed increases until speeds of 3,600 ft. per minute and upwards are reached, when the influence of the speed is no longer felt.

It seems safe to say that with a given bearing that has been carefully fitted and "run in" to a proper bearing surface, the friction involved will be entirely viscous friction as long as the oil film is maintained. It follows, therefore, that all oils of the same absolute viscosity will offer the same frictional resistance under this condition. In other words, their coefficient of friction will be equal for identical conditions of operation in the same bearing. Hence the lubricating value of a given oil will depend only on its viscosity characteristics over the operating temperature and pressure range encountered in that particular bearing. Needless to say oils have other properties which fix their suitability for a given purpose which are external to their lubricating qualities, such as evaporation, oxidation, emulsification, sludging, etc.

An additional factor is introduced, however, when we consider that in the cycle of operation the "oiliness" of the lubricant comes into play when the machine is being started and again when it comes to rest and there is metal to metal friction even though this period is of short duration. It is also considered desirable to have one kind of lubricant to serve all conditions of lubrication, that is, the oil that is used on the main units would also be used on external parts that are oiled by hand and on the auxiliary machinery in which the lubrication is characteristically of the intermediate kind.

It appears, therefore, that although viscosity plays the leading role where film lubrication is maintained, yet it would be desirable to have an "oily" lubricant for the reasons outlined above, provided the admixture of the elements necessary to accomplish this did not impair the efficacy of the lubricant in maintaining its film. The question to be considered, therefore, is whether a straight mineral oil that has been treated with fatty acid to increase its oiliness maintains the oil film under critical conditions of load and speed as well as the untreated oil.

It was found by test with the oils used in this work that the addition of oleic acid up to two percent had no apparent effect on the viscosity at atmospheric pressure. It is very probable, however, that under higher pressures this would not be the case as it is known that the viscosities of fixed oils under pressure do alter materially as well as the viscosities of the mineral oils. In oils treated with oleic acid, one would expect to find a condition somewhere between the two.

To recapitulate, consider a bearing with fixed characteristics supplied with mineral oil of known viscosity. After the machine has been started there will be a short period in which metallic contact friction is involved. This condition will endure until a speed is reached sufficient to form and maintain the fluid film. When this condition obtains, the friction will depend entirely on the viscosity, of course, taking into

consideration the fact that the viscosity changes greatly with temperature and somewhat with pressure. As the machine is brought up to speed the friction will increase with the speed, decrease with rise in temperature and decrease as the concentric position of the journal is approached.

Now with a given speed and increased load the friction will increase with pressure and eccentricity of the journal until there will be a point reached wherein the film of lubricant will be so thinned that there will be momentary contacts between the metallic surfaces. When this condition of operation is reached the friction involved will be both fluid and contact. It will be principally fluid friction at first but as the pressure is increased and the areas of contact grow, it gradually changes into contact friction until finally a critical pressure will be reached at which the film will be entirely displaced. When this happens, seizing of the entire bearing area occurs and if motion of the parts is maintained abrasions of the bearing surfaces will take place.

With the main features of the above discussion in mind it is apparent that in bearing design the critical pressure causing break-down of the oil film must be known in order to proceed rationally. It is safe to say that in all oils this critical pressure under precisely similar conditions of operation depends largely upon the viscosity but the question naturally arises whether this is specifically true of all kinds of oils of the same viscosity, particularly of the mineral oil treated with free fatty acids.

Statement of the Problem.

This brings us to a point where the problem can be stated in exact terms, to wit:

> To determine the critical or break-down pressure in the oil films formed in a bearing, using straight mineral oils and also to determine the influence on this critical pressure of the admixture of definite amounts of oleic acid.

Previous Work.

Little experimental work has been done with the definite object of determining these break-down pressures. The usual method of attack follows the lead of Sommerfeld (Zeit für Math. u. Phys. 1904, 597) who made use of the experimental results of Stribeck (Mitteilungen Uber Forschungsarbeiten, Heft. 7, 1903).

An examination of the curves in Sommerfeld's paper (Zeit. für tech. Physik, Vol. 2, 1921) discloses the fact that when pressures are plotted as abscissa and coefficient of friction as ordinates on rectangular coordinates, the coefficient of

friction decreases with increase of pressure until a certain minimum value is reached, but beyond this point it increases. This minimum value of the coefficient of friction has the same magnitude for all velocities of bearing face. The curves, however, present distinctly different trends with various values of velocity. At low speeds the slopes are very abrupt but flatten out as the speeds are increased. This minimum, or "transition point", as it is called, is regarded as being at or near the break-down point of the oil film. This seems reasonable, if the notion is valid that there is a gradually increasing area of metallic contact after the critical pressure is reached. In that case it would certainly tend to increase the friction above that which could be attributed to viscous friction, in proportion to the increasing area of the contact surfaces.

Sommerfeld gives the following equations for the condition at the "transition point" for a bearing completely surrounding the journal:

$$\mu = \frac{50 \quad P}{\pi^2 \; N} \left[\frac{\Delta}{d}\right]^2 \qquad (9)$$

where
μ = viscosity in poises
P = pressure in dynes per sq. cm. of projected area
N = revolutions per minute
Δ = difference in radii of bearing and journal in cm.
d = diameter of journal in centimeters.

If the viscosity is expressed in poises, the pressure must be stated in dynes per square centimeter, or 68,9[illegible].5 times the pressure in pounds per square inch, so that if it is desired to use English units throughout, equation (9) may be written thus:

$$\mu = \frac{349,556.5 \; P}{N} \left[\frac{\Delta'}{d}\right]^2 \qquad (10)$$

or

$$P = \frac{\mu N}{349,556.5} \left[\frac{\Delta'}{d}\right]^{-2} \qquad (11)$$

where
μ = viscosity in poises
P = pressure in pounds per sq. in. of projected area
N = revolutions per minute
Δ' = thickness of oil film in inches
d = diameter of journal in inches

Equation (11) is a very simple relationship and would be very convenient if it were not for the difficulty in measuring the thickness of the oil film Δ'. This difficulty is a very real one and nullifies this line of attack, especially in view of the fact that after actual rupture of the film has started Δ' becomes zero at some points and of course it is impossible to say where contact occurs or what is the extent of this area of contact.

Sommerfeld's equation [illegible] deduced on the [illegible] the bearing completely surrounded the journal [illegible] be hazardous to contend that the same relation would hold good with a bearing having an arc of contact of 120° or even less [illegible] usually [illegible] in practice.

Dr. J. T. Nicolson in a very able paper (Proc. Manchester Assn. of Engineers, Nov. 1[illegible]) following a similar method of deduction, after a very careful analysis arrived at the following equation for various cases usually occurring in practice:

$$P = 40 \left[d \quad N \right]^{\frac{1}{4}} \qquad (12)$$

where
- P = pressure in lbs. per sq. in. projected area
- N = R. P. M.
- d = diameter of journal in inches

Formula (12) is intended for the design of bearings which completely surround the journal and, of course, in that respect has the same limitations as that of Sommerfeld but nevertheless if the premises from which it is derived are correct it should yield a value which is near the critical pressure.

The actual measurement of lubricating film thicknesses has been reported by Kingsbury (A. S. M. E., May 1897), Green (Jour. A. S. M. E., April 1917) and Stoney (Engineering, [illegible] Mar. 192[illegible]) but the only report of a serious attempt to determine the actual break-down pressure of the film in a quantitative sense is that of Prof. Herbert S. Moore (Amer. Mach. Vol. [illegible], p. 1[illegible]81).

Prof. Moore's experimental apparatus consisted of a test journal having a diameter of 1.9[illegible]6 inches, arranged so that the lower half would revolve in an oil bath. To this journal was fitted a half bearing so contrived that the load could be varied by applying known weights to a hanger attached to the bearing. The journal and bearing with the oil film between them formed part of an electric circuit. When the machine was in operation and the oil film formed, this film would offer a great resistance to the flow of current, while on the other hand if the film were broken this resistance would fall off to zero or thereabouts. A voltmeter was connected across the line to indicate the line potential and in this way the pressure causing rupture of the film could be determined.

With the data obtained in this way Prof. Moore plotted a curve of speed in feet per minute as abscissa and break-down pressure in pounds per square inch of projected area as ordinates. He found that the following expression fitted this curve very well over the range of his observations:

$$P = 7.47 \sqrt{V}$$

where
- P = break-down pressure in pounds per sq. in. of projected area.
- V = rubbing velocity in feet per minute.

This equation has been largely used in design. Mr. Axel K. Pedersen writing in the Amer. Mach. 10 Oct. 1912 says: "This equation is fundamental and is now considered as a very close approximation by the best authorities, and should, therefore, be used in all cases where a perfect oil-film lubrication is desired".

Dr. Sanford A. Moss, Chief Engineer of the Mechanical Research Division of the General Electric Company, in a communication to the writer refers to Prof. Moore's work as "a classic in the matter" and suggested that a check and extension of Prof. Moore's work would be highly desirable.

It was with this end primarily in view that the present investigation was undertaken. It was thought desirable, however, to devise some mechanical means of determining the break-down pressure and using the electrical method as a check. The first method contemplated was to fit a solid bearing with known clearance to a journal supported by two pedestals and load the bearing by means of a lever and jack. It was proposed to fit a torsion meter to the shaft and determine the break-down of the film by the sudden increase of torque indicated on the torsion meter. This plan was abandoned principally on account of the time required to build the apparatus and also because an optical torsion meter, which was thought essential to this arrangement, was not immediately available. The apparatus finally decided upon can best be illustrated by the detailed description which follows:

Description of Experimental Apparatus.

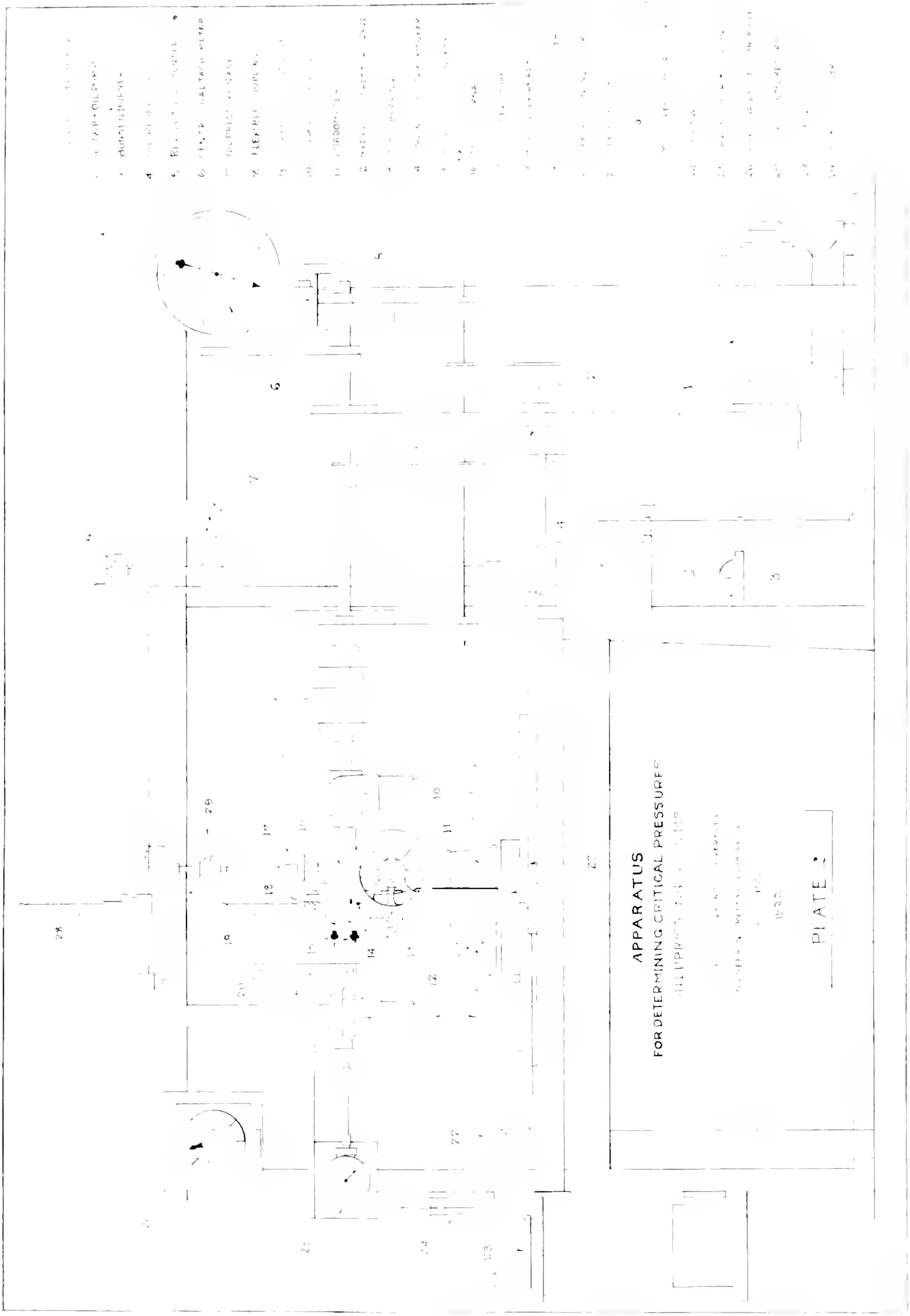
In the following description of the apparatus it will be convenient to refer the numerals to the diagram shown in Plate 2 and locate the parts described on the photograph of the set-up shown in Plate 1.

The experimental apparatus was constructed in the university shop and was made entirely of miscellaneous equipment found about the laboratory. The design was influenced largely by what was available rather than by what was thought to be desirable.

The machine consists essentially of the test journal (16) which is made of Sanderson's tool steel, manufactured by the Crucible Steel Company of America and is described by that company as a straight carbon tool steel, containing about one percent of carbon, with a phosphorous and sulphur content under 0.025 percent. The journal was turned oversize, hardened and then ground and lapped to exactly two inches diameter.

The journal is supported in its pedestal by two bronze bushed bearings (10). The pedestal is firmly bolted to a heavy oak laboratory table and this table is bolted to the frame that supports the driving mechanism by heavy timbers.

PLATE I


APPARATUS
FOR DETERMINING CRITICAL PRESSURES
PLATE 2

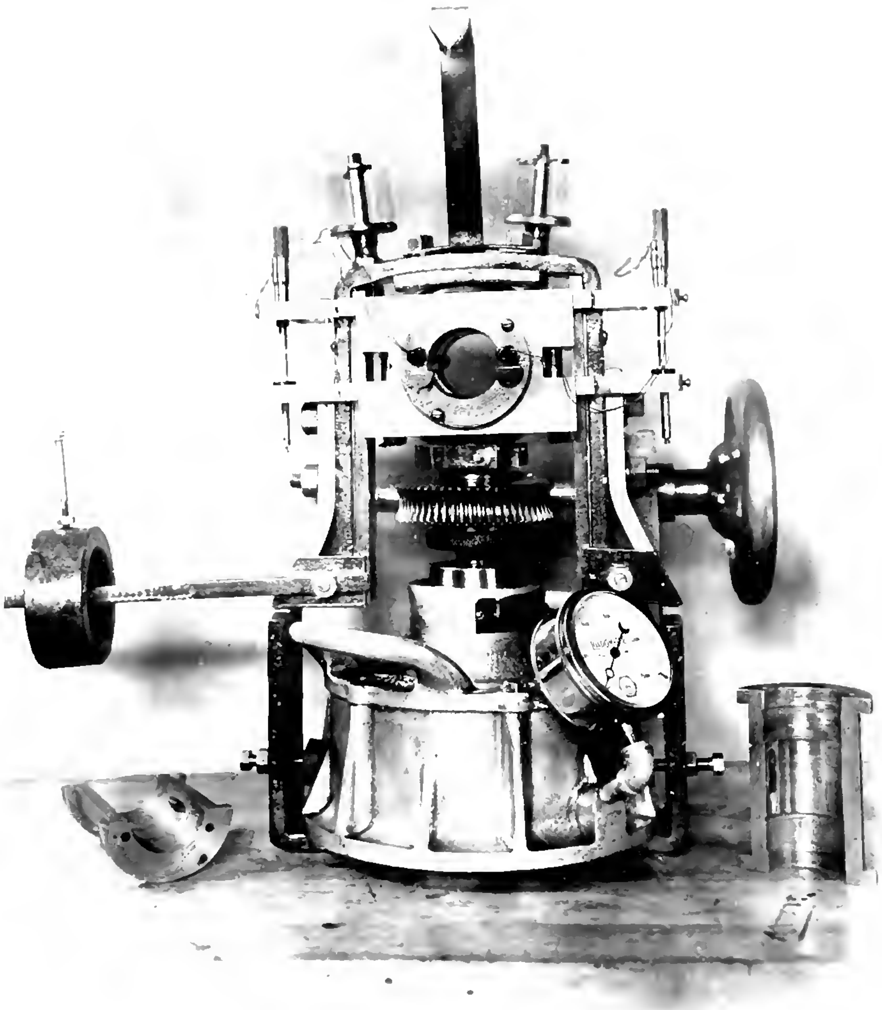


FIG 1

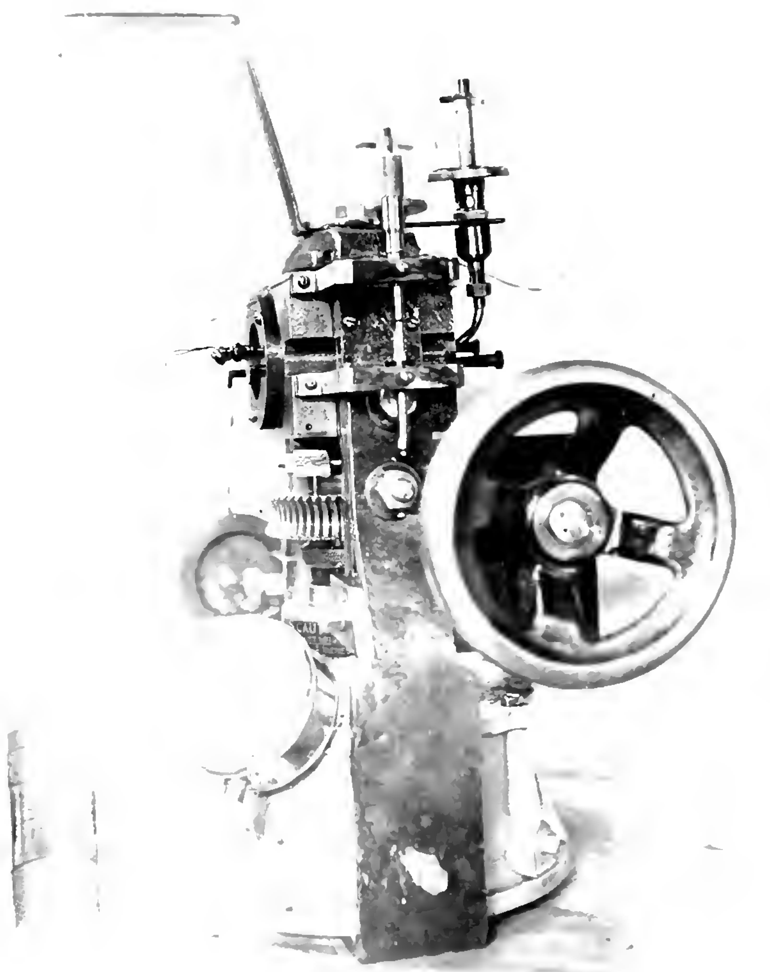


FIG 2

PLATE 3

The machine is belt-driven by a 3 H. P., 1200 R. P. M., variable speed, shunt motor and is controlled by the starting panel and controller (25). Close graduation of the speed is secured by the armature resistance (26). Alignment of the driving shaft is maintained and transmitted vibration from the motive element is reduced by a Fast flexible coupling (8), manufactured by the Bartlett-Hayward Company of Baltimore.

The test bearing consists of a split bearing (13), the two halves of which are pressed together on the journal by means of a hydraulic loading device (11). A suitable steel strap surrounds the jack and bearing so that the thrust of the jack will be applied to both halves of the bearing. This loading device is known as a Loadometer and is manufactured by the Black and Decker Company of Baltimore. The total load applied by the jack is indicated by the pressure gage (12) which is graduated to read in pounds. The load is applied by rotating the worm gear by means of the hand wheel (14).

The bearing with the jack, loading mechanism, supporting strap and other appendages, constitute a pendulum, the displacement of which through any angle indicated by the pointer (20) on the scale (21), which is graduated in degrees, will be a measure of the friction. The total friction and the coefficient of friction can be determined as follows:

Let W = Weight of pendulum, (114 pounds).
R = Radius from center of journal to center of gravity of pendulum, (6 inches).
r = Radius of journal, (1 inch).
P' = Total load on bearing.
ϕ = Angular displacement of the pendulum from the vertical.
f = Coefficient of friction.
F = Total frictional resistance.

The total load P' is the value indicated by the gage, plus the Loadometer calibration correction and the weight of the pendulum. This is sufficiently accurate in this case as the total loads encountered are large in proportion to the weight of the pendulum, but it should be borne in mind that the load on the upper half of the bearing is greater than that on the lower by an amount approximately equal to the weight of the pendulum.

When $\phi = 90^\circ$, $W R = F r$. Then for any other angle, $W R \sin\phi = F r$. The expression for the coefficient of friction may be written:

$$f = \frac{F}{P'} \quad \text{but} \quad F = \frac{W R \sin\phi}{r}$$

then

$$f = \frac{W R \sin\phi}{P' r} = \frac{114 \times 6 \sin\phi}{P'}$$

$$f = \frac{684 \sin\phi}{P'}$$

With this, a tabl of coefficients of friction could be calculated for a wide range of deflections and pressures.

It should be noted that the radius R in the above expressions is the same at all conditions of load, as the movement of the piston is very small indeed, so small, that R is not disturbed under varying conditions of load. This feature is unique and is considered a notable improvement over the spring loaded machines in which the center of gravity changes with every change of load.

The actual bearing surface is made in the form of a split sleeve which is bolted in the cast-iron holder. The cast-iron sleeve is shown in Plate 7, Figs. 1 and 2. This sleeve carries the bearing surface which on the first trials was made of close-grained cast-iron and on the later of high grade babbitt metal, manufactured by the United American Metals Corporation of Brooklyn, N. Y. and described by that company as Syracuse Government Genuine Babbitt, containing approximately 90 percent tin, 3 percent copper and 7 percent antimony.

The bearing block which carries the bearing sleeve is split in the same manner as the sleeve and provided with spherical sections at the top and bottom that fit into similar female sections attached to the strap at the top, and the head of the jack at the bottom, so that the bearing is self aligning.

The oil supply is maintained by a rotary pump (2) which takes its suction from the reservoir (4) and discharges into the sight feed oil cups (17) and by this means a copious supply of lubricant is assured. The oil inlet line is vented by the valve (9) so that entrained air will be discharged through it, and not carried into the oil supply. This valve also serves to regulate the pressure on the oil line which is indicated by the gage (7). The oil is carried to the sight feed cups by means of the flexible metallic tubing so there will be no resistance to the movement of the pendulum at this point. The oil return is accomplished by means of the drip-pan and trough (27) which act also as an oil cooler.

The speed is measured by an Elgin Chronometric Tachometer (25) suplemented by a centrifugal tachometer (6) and a positive counter (5). The Elgin Tachometer functioned perfectly and proved to be highly satisfactory for the work. No irregularity could be detected by comparing it with the counter.

The temperature of the bearing was determined by a mercurial thermometer (19) which was supported by a fiber bushing in a recess drilled into the bearing within an eighth of an inch of the bearing surface. Thermo-couples made of copper and "Advance" wire were fitted into recesses right at the entering and leaving edges of the actual bearing area so that the oil temperatures at both could be determined. The cold junctions of the thermocouples were immersed in a thermos bottle and the current generated was determined by a highly sensitive galvanometer so that small differences in temperature could be detected.

Vernier micrometers (18) that could be read to ten thousandths of an inch were fitted to each half of bearing supporting blocks so that the vertical movement of the two halves of the bearing could be measured in relation to each other and in this way an approximate measure of the film thickness determined. The method of fitting these micrometers is shown in Plate 3, Figs. 1 and 2.

Operation of the Machine.

The method of operation is as follows: The machine is started by closing the line switch (24) and bringing the motor up to speed by the controller (23). The load is then applied by the hand wheel (14) which actuates the hydraulic loading device. The application of the load is accomplished by pressing together the two halves of the test bearing (13) which are free to move in the supporting strap but are constrained to move vertically in relation to each other by hand fitted dowel pins which run through both halves of the bearing supporting block.

As the pressure increases the oil film thins out until small points of contact occur. Before this point is reached the pendulum remains perfectly steady and the pointer (20) indicates a definite deflection which is a measure of the fluid friction, beyond this point, however, the pendulum grows increasingly unsteady until the critical point is reached, and the bearing grips the journal. When this occurs the pendulum is suddenly deflected through a large arc until its weight is sufficient to cause the driving belt, which is loosely fitted, to slip. The operator then opens the cut-out switch (22) and shuts down the machine.

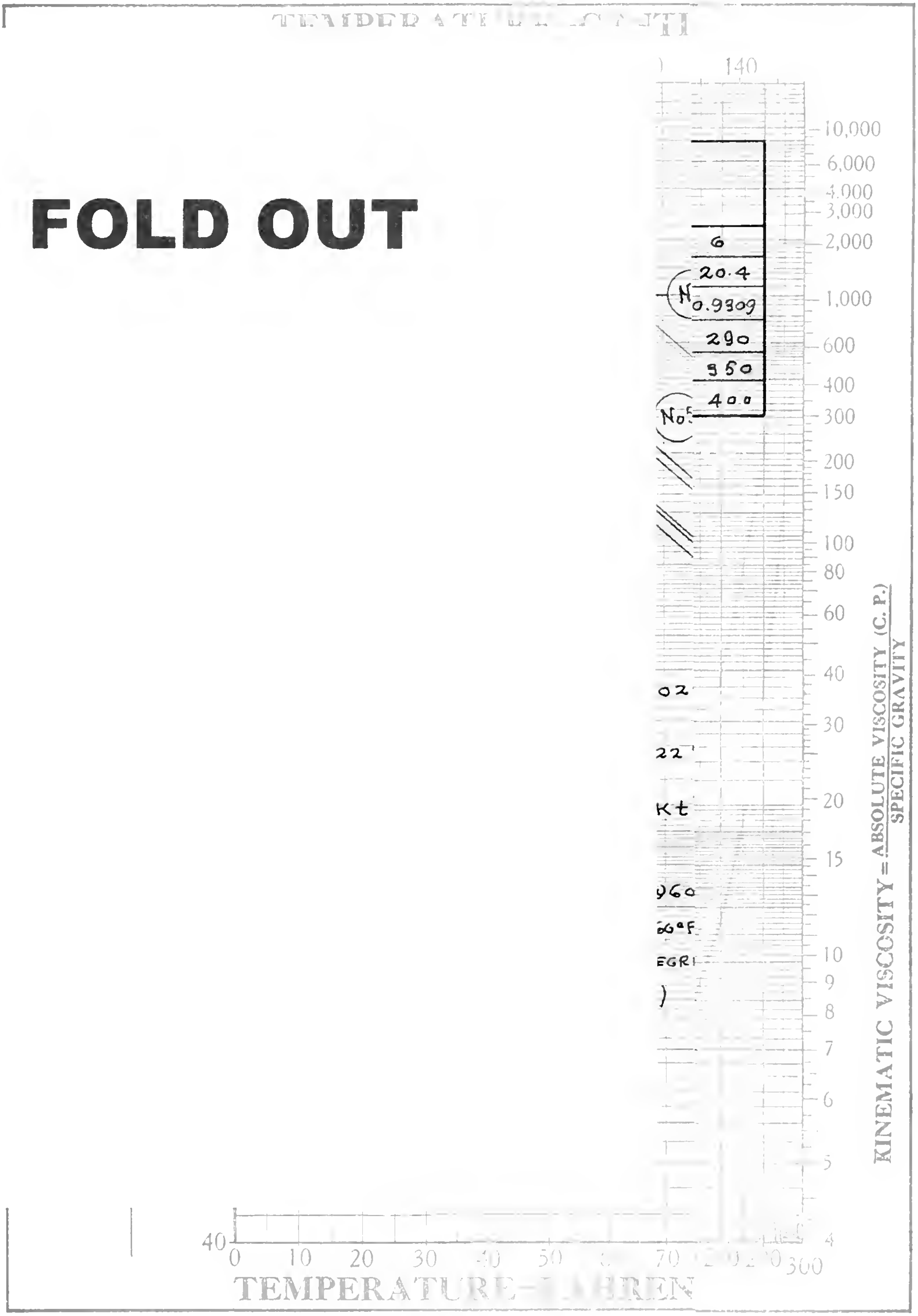
In getting the true break-down pressure any irregular mechanical application of force is avoided by bringing the pressure up to a value just below the critical point and then allowing the increase of temperature to cause the viscosity to decrease until rupture occurs.

Oils for Test.

The oils for the test were supplied through the courteous cooperation of Dr. Raymond Haskell of the Texas Company, and were procured direct from the refinery at Port Arthur, Texas, so there would be no chance of them becoming contaminated by the admixture of unknown ingredients.

The characteristics of these oils, as determined by the company's chemist, are given in Table No. 1. The viscosities and gravities were carefully checked in the laboratory at the University.

FOLD OUT



[VISCOSIT

TABLE No. 1

	A	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6
Gravity Bé.	21.6	21.4	20.2	26.0	24.8	20.8	20.4
Spec. Gravity	.9235	.9247	.9321	.8974	.9044	.9284	.9309
Flash	340	330	385	375	410	355	350
Fire	400	385	460	450	470	410	400
Viscosity @ 100° F. (Saybolt)	202	205	765	162	750	708	290

Oil A is a commercial red oil (200 Saybolt) and oils 1, 2, 5 and 6 are of Gulf Coast origin, while 3 and 4 are from Mid-Continental fields. All of these oils are straight mineral oils without compounding or blending.

The Saybolt and kinematic viscosities of all these oils covering the range of temperature used in the experiments are plotted on Chart No. 1. The form of this chart was devised by Mr. MacCoull of the Texas Company and is very useful for the purpose. The absolute viscosities of oils A, 1 and 5 which were used in the final runs, are plotted on Chart No. 2.

Commercial oleic acid was used in the fatty acid treatment of these oils and contained 91 percent oleic acid, the rest being fixed oils, principally lard oil.

Preliminary Trials.

In the preliminary trials a cast-iron bearing, having a length of four inches was used with oil A of Table No. 1. It was apparent immediately that the apparatus would function as intended but just as soon as the bearing began to seat itself the pressures necessary to break the film became so great at high speeds that they taxed the capacity of the machine, so it was decided to cut the bearing length down to two inches.

Trials with Cast-Iron Bearing.

The original bearing was modified by milling off the excess bearing surface to a length of two inches. It was first carefully fitted by hand then lapped in on an arbor with a lapping compound composed of 46 percent silica, 40 percent white lead, 7 percent sodium carbonate, 4 percent black lead and 3 percent fullers earth. This method produced a very fine bearing surface but in addition the bearing was "run in" under some pressure before the actual test runs were started.

The final trials with the cast-iron bearing were also run with oil A. The results are given in tabular form in Table A and graphically on Chart A. The numerals near the plotted points indicate the absolute viscosity of the oil in poises. On this chart is also given the curves determined from Dr. Nicolson's equation and also from Prof. Moore's.

TABLE II.

Test with Cast-Iron Bearing.

Using Oil A.

Speed in Feet per Minute V	Break-Down Press. in Lbs. per Sq. In. Proj. Area P	Absolute Viscosity in Poises μ
15[illegible]	1[illegible]	[illegible]
15[illegible]	1[illegible]0	0.[illegible]
[illegible]	1[illegible]0	[illegible]
[illegible]5	1440	0.[illegible]
[illegible]5	15[illegible]	[illegible]
[illegible]	1700	0.[illegible]1
[illegible]10	1140	[illegible]
[illegible]0	1[illegible]0	0.1[illegible]
350	15[illegible]	[illegible]1
[illegible]0	1[illegible]	0.14
[illegible]	14[illegible]0	0.17
41[illegible]	1[illegible]00	0.1[illegible]
410	1540	0.1[illegible]

Speed in Feet per Minute V	Break-Down Press. in Lbs. per Sq. In. Proj. Area P	Absolute Viscosity in Poises μ	
[illegible]50	[illegible]	.[illegible]4	x
4[illegible]	[illegible]40	.1[illegible]	x
[illegible]	10[illegible]0	.[illegible]1	x
[illegible]45	1140	0.[illegible]4	x
4[illegible]	1400	0.[illegible]	
4[illegible]5	1540	0.[illegible]0	
5[illegible]	1640	0.14	
5[illegible]	630	.[illegible]5	x
5[illegible]0	1460	0.1[illegible]	
[illegible]75	1400	0.12	
575	1650	0.1[illegible]	
5[illegible]	1140	0.1[illegible]	x
615	1400	0.1[illegible]	

Points marked x due to lint in bearing.

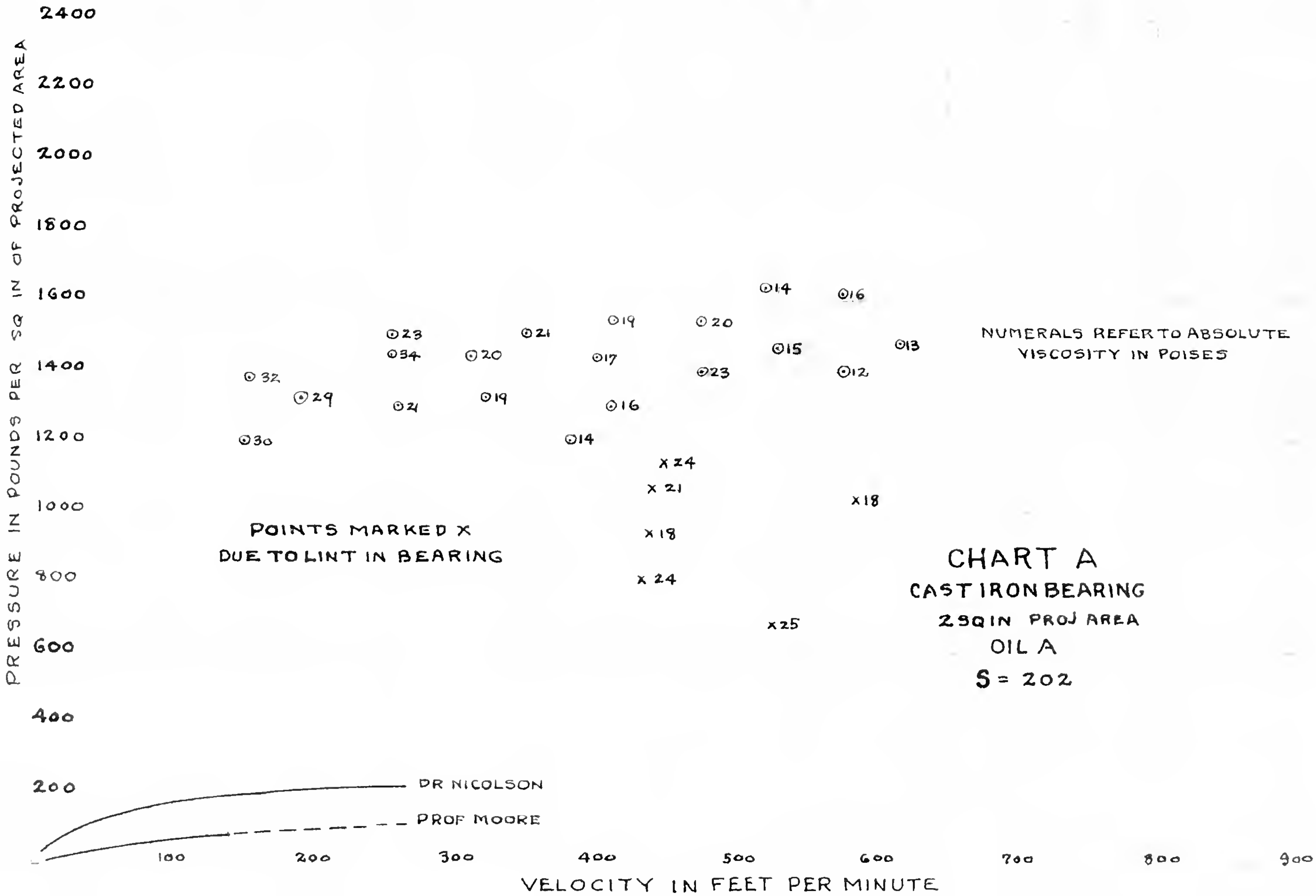


CHART A
CAST IRON BEARING
2 SQ IN PROJ AREA
OIL A
S = 202
NUMERALS REFER TO ABSOLUTE VISCOSITY IN POISES
POINTS MARKED X DUE TO LINT IN BEARING
DR NICOLSON
PROF MOORE
PRESSURE IN POUNDS PER SQ IN OF PROJECTED AREA
VELOCITY IN FEET PER MINUTE
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
100
200
300
400
500
600
700
800
900

Three important points were brought to light in these trials, namely:

(a) That the pressure causing rupture of the oil film was much greater than that calculated by either Dr. Nicolson's (18) or Prof. Moore's (17) equations.

(b) That with each refitting of the bearing the pressure required to break the film was higher and even without a refitting, but after the bearing seated itself better, the rupturing pressure grew higher.

(c) That in spite of great care taken to remove it, the lint from the rags which were used in wiping out the drip pans, return trough and reservoir would collect in a solid wad at the entering edge of the bearing surface and cause in consequence very low and discordant break-down points. This fault was remedied by fitting 50 mesh copper wire strainers to the end of the oil inlet tubes.

Although the results of these trials were surprising, no great importance was attached to the value of the break-down pressure obtained as it became apparent later that the real relationship was obscured by mechanical crudities of operation and by the lack of permanence in the fit of the bearing.

It was found after some runs at high speeds that the seizing was causing injury to the journal and it was decided to go no further with the cast-iron bearing, but to shift over to the babbitt one. The abrasion of the journal was very slight and it was successfully lapped out with the lapping compound previously mentioned. This lapping process reduced the diameter of the journal about four ten thousandths of an inch. After this had been done, no unevenness could be detected along the bearing surface.

<u>Trials with Babbitt Bearing.</u>

A special cast-iron sleeve was made but turned one sixteenth inch large, recessed and undercut, to hold the babbitt metal liner of two inches length. The babbitt was poured in the usual manner, peened and then turned a little oversize. It was then fitted by hand and lapped in the same manner as the cast-iron bearing but in addition was touched up finally with powdered Sapolio on the test journal. This method produced a very fine bearing surface which would become highly polished when in use a short time, but it was found to be impossible to make this polished contact surface occupy the entire bearing surface available. The actual bearing area would grow larger with each refit and

with wear, but even when the experimental work was stopped only about 80 percent of the bearing surface had come into play. It is to this fact that the progressively higher pressures with each refit is attributed. Entirely concordant results could only be expected after an unchanging, highly polished bearing surface had been produced. It is thought that to produce such a surface, covering the entire available area, it would take weeks, perhaps months of "running in" the bearing under considerable pressure.

The babbitt used was evidently of a very superior quality, as it was found that no abrasion of the bearing surface or the journal occurred, even under the most severe conditions of operation.

The preliminary trials with the babbitt bearing and oil A indicated immediately that the break-down pressures would be much higher than those encountered with the cast-iron bearing. After the second refitting of the bearing the break-down pressures became so high that the 3/8 inch steel tap bolts, which held the strap together, were repeatedly sheared and in fact the strap itself was distorted to such an extent that it looked as if it might carry away any minute.

It was obvious that as the bearing seated itself better the break-down pressures would go quite beyond the strength of the strap, so it was decided to forge a new strap in a solid piece thus eliminating the tap screws. It was also decided to reduce the projected area of the bearing to 2.764 square inches.

When these two modifications had been made the bearing was again refitted and lapped in as previously described and then "run in" with oil A.

The systematic testing then began, using in succession, in the order named, oils A, No. 1, No. 2, No. 3, No. 5, No. 4 and No. 6. Oils No. 2 and No. 4 were so viscous and the flow conditions with them so poor that the tests with them were abandoned. Repeated runs with the other oils confirmed the fact that the break-down pressures were many times greater than those which had heretofore been regarded as probable. It developed as the tests progressed, that in each succeeding run the break-down pressure grew higher, confirming the conclusion that had already been reached that the fit was the controlling factor. In any period, however, fairly consistent results were obtained and one could predict quite closely at what point the rupture of the film would occur.

Unfortunately, when the bearing was reassembled after examination and a new trial started, the former relationship would be disturbed considerably.

No very close agreement with the former values could be attained, but as the work progressed it became more evident that as the fit improved the results were more consistent.

Final Trials.

After the trials had progressed to such an extent that the results grew consistent to a fair degree, it was decided to proceed with the final trials. Oils No. 1 and 5 were selected as their viscosities covered the range ordinarily encountered in practice.

The procedure in this case was to run a series of trials with the straight mineral oils until enough data had been determined, then shift over to the treated oils and repeat the process.

The results of these trials are given in Tables B and C, and shown graphically on Charts B and C.

Comment on Curves.

The curves drawn on the Charts B and C are the same and are not intended to be curves that represent the characteristic behavior of the break-down pressure. They are drawn merely to aid the eye in following the trend of the data. It would be too much to say that the curves represent what really happens, because it is recognized that the machine was mechanically deficient, the degree of fit not perfect, and the temperature observed was probably not that which obtained in the oil film itself. It is believed, however, that the data recorded does sensibly represent the magnitude of the break-down pressures with that particular degree of fit, but with another refitting of the bearing the pressures would certainly go still higher and the trend of the curves might be disturbed entirely.

Contrary to expectation the break-down pressures obtained after the oils had been treated with the 2 percent oleic acid were materially less than with the untreated oils. The break-down point was not so sharply defined as with the straight mineral oils. The unstable period which indicated contact was much longer and in consequence the actual critical points were very erratic. This apparent discordance was no doubt due in a large measure to the fact that the real temperature of the film was unknown as in this case the contact friction caused the temperature to rise very rapidly and no doubt it was not properly registered by the method used.

The plotted data indicate that the break-down pressure does depend largely on the viscosity as an inspection of the curves show that although there is no great degree of concordance there is certainly a marked similarity between the two sets of data.

TABLE B.

Test with Babbitt Bearing.

Using Oil No. 5.

Untreated

SPEED IN FEET PER MINUTE	BREAK-DOWN PRESS. IN LBS. PER SQ. IN PROJ. AREA	OBSERVED TEMP. IN DEGREES F.	ABSOLUTE VISCOSITY IN POISES
V	P	t	μ
155	3500	121	0.32
160	3750	120	0.33
180	3250	130	0.26
200	3100	148	0.17
205	3600	129	0.27
230	3150	154	0.15
250	3650	133	0.24
270	3850	130	0.26
285	3200	175	0.10
290	3450	154	0.15
310	4200	124	0.30
325	4050	129	0.27
325	3700	142	0.19
330	3300	175	0.10
330	3950	133	0.24
335	3500	160	0.13
340	4100	156	0.29
365	3750	145	0.18
370	3900	136	0.22
390	4200	127	0.22
395	4350	127	0.28
425	3750	151	0.16
435	3850	143	0.17
435	3650	157	0.14
445	4350	129	0.27
450	4050	138	0.21
400	3450	170	0.11
475	4400	129	0.27
495	3600	165	0.13

Untreated

SPEED IN FEET PER MINUTE	BREAK-DOWN PRESS IN LBS. PER SQ. IN. PROJ. AREA	OBSERVED TEMP IN DEGREES F	ABSOLUTE VISCOSITY IN POISES.
V	P	t	μ
495	4050	145	0.19
500	4200	130	0.22
530	4400	132	0.25
530	3650	165	0.12
535	3950	155	0.19
550	3750	157	0.14
555	3900	154	0.15
565	3500	190	0.09
575	4150	139	0.20
575	4050	142	0.10

Treated

V	P	t	μ
200	2900	118	0.36
215	2450	121	0.32
260	3200	117	0.37
275	2900	139	0.20
335	2900	151	0.16
375	3200	145	0.15
400	2750	157	0.14
410	3100	151	0.16
460	3050	170	0.11
475	3600	132	0.21
490	3300	157	0.14
500	2800	165	0.12
535	2950	160	0.13
575	3250	157	0.14

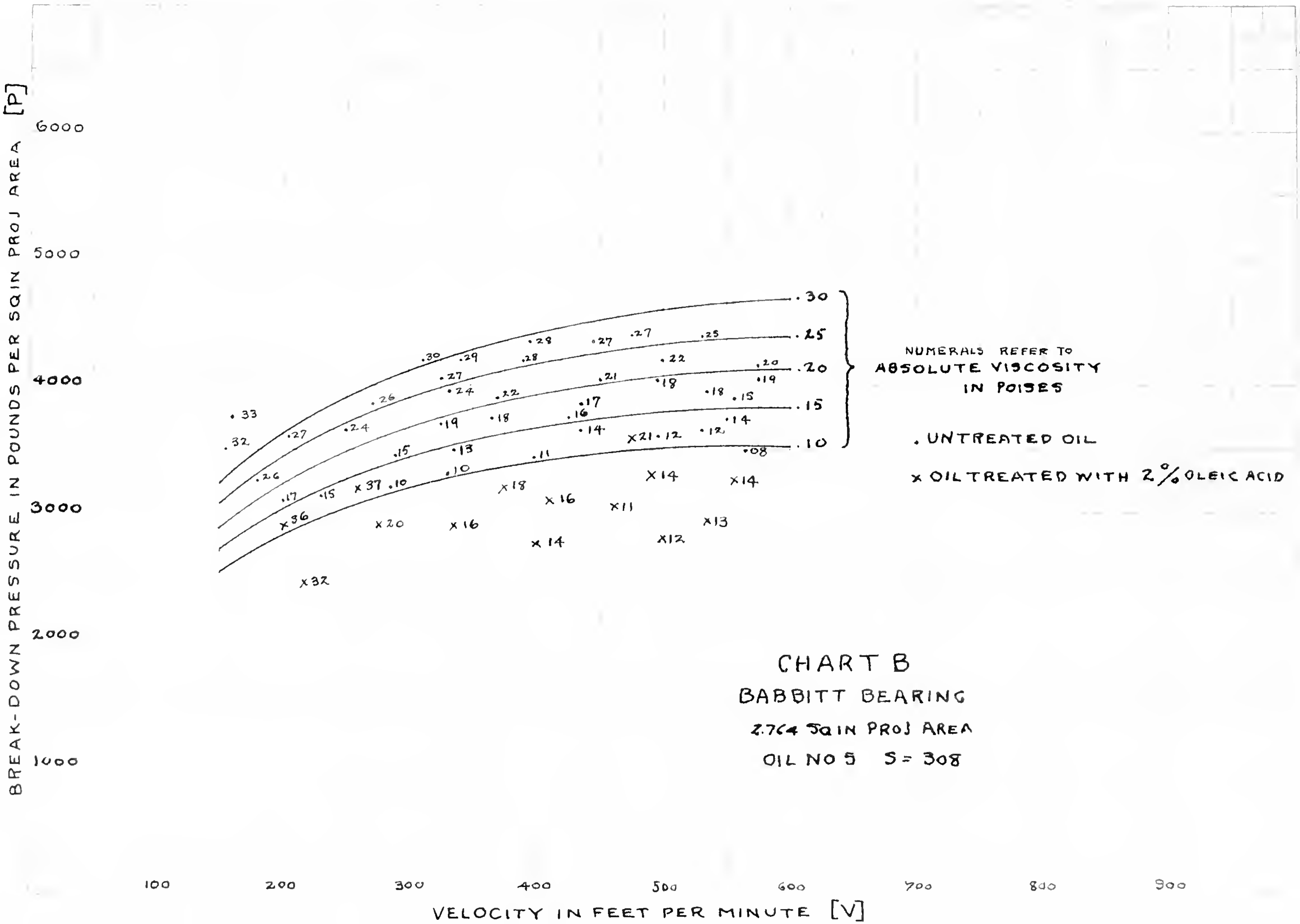


CHART B
BABBITT BEARING
2.764 SQ IN PROJ AREA
OIL NO 5 S = 308
NUMERALS REFER TO ABSOLUTE VISCOSITY IN POISES
. UNTREATED OIL
x OIL TREATED WITH 2% OLEIC ACID
BREAK-DOWN PRESSURE IN POUNDS PER SQ IN PROJ AREA [P]
VELOCITY IN FEET PER MINUTE [V]
1000
2000
3000
4000
5000
6000
100
200
300
400
500
600
700
800
900
.30
.25
.20
.15
.10

TABLE C.

Test with Babbitt Bearing.

Using Oil No. 1.

Untreated

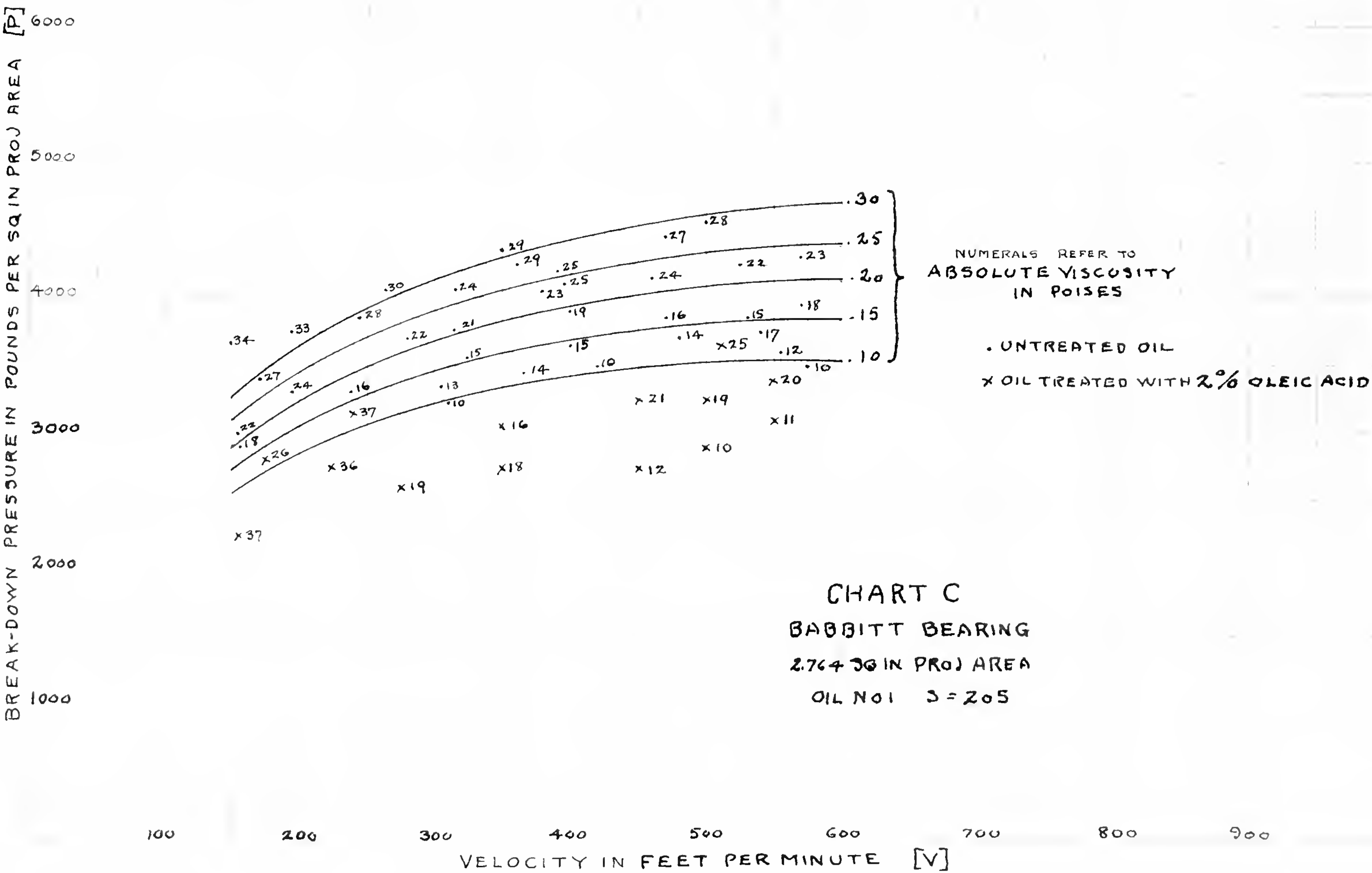
Speed in Feet per Minute V	Break-down Press in Lbs. per Sq. In. Proj. Area P	Observed Temp. Degrees F. t	Absolute Viscosity in Poises μ
150	3650	107	0.34
155	3000	123	0.22
155	2000	172	0.13
170	3400	115	0.27
195	3750	108	0.32
195	3500	119	0.24
240	3500	177	0.16
245	3850	114	0.28
265	4050	112	0.30
280	3700	113	0.22
305	3350	148	0.13
310	3250	162	0.10
315	4050	119	0.24
315	3750	135	0.21
325	3550	140	0.15
350	4350	112	0.29
360	4150	113	0.29
365	3450	144	0.14
380	4050	120	0.23
390	4200	116	0.25
395	4100	118	0.25
400	3200	130	0.19
400	3650	140	0.15
420	3500	163	0.10
460	4150	110	0.24
470	4450	115	0.27
470	3350	137	0.16

Untreated

Speed in Feet per Minute V	Break-down Press in Lbs. per Sq. In. Proj. Area P	Observed Temp. Degrees F t	Absolute Viscosity in Poises μ
400	3700	144	0.14
500	4550	114	0.[illegible]
525	4050	127	0.[illegible]
530	3650	141	0.15
540	3750	135	0.17
555	3000	159	0.12
570	4300	120	0.23
570	3950	132	0.18
575	3500	163	0.10

Treated

V	P	t	μ
150	2250	104	0.37
175	2800	117	0.[illegible]
215	2750	105	0.36
240	3150	104	0.37
275	2600	150	0.1[illegible]
350	3050	137	0.10
350	2750	132	0.18
450	3050	135	0.[illegible]1
450	2750	132	0.12
500	3250	130	0.19
500	2900	162	0.10
510	3650	112	0.25
550	3400	137	0.20
550	3100	157	0.11



NUMERALS REFER TO ABSOLUTE VISCOSITY IN POISES
. UNTREATED OIL
x OIL TREATED WITH 2% OLEIC ACID
CHART C
BABBITT BEARING
2.764 SQ IN PROJ AREA
OIL NO I S = 205
BREAK-DOWN PRESSURE IN POUNDS PER SQ IN PROJ AREA [P]
VELOCITY IN FEET PER MINUTE [V]

Electrical Method of Determining Break-down Pressure.

The electrical method of determining the break-down pressure was tried during the trials with the cast iron bearing. The arrangement was similar to that used by Prof. Moore. It consisted simply in connecting the bearing and journal in a circuit as shown in Fig. 2.

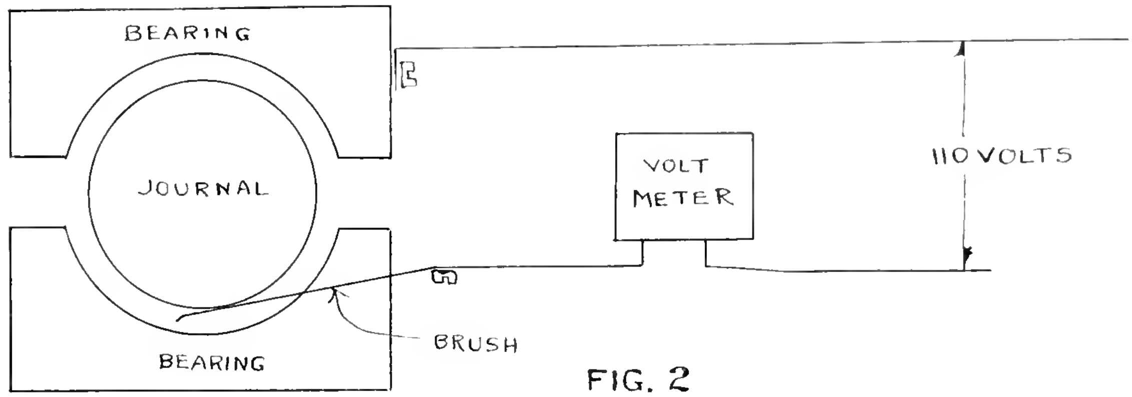


FIG. 2

The theory is, that as long as the film was maintained, the voltage, indicated on the voltmeter, would be zero but when the film was ruptured, metallic contact occurred and the voltmeter would indicate full line voltage.

In the first attempts with this method the apparent break-down pressures were so far below that indicated by the mechanical method, it was thought that perhaps the metallic particles that were being stripped from the bearing and carried along in the oil was responsible. The attempt was discontinued at that time but was tried again in the final trials.

These second attempts were not very much more successful than the first; the apparent break-down occurring at a few hundred pounds when it was obvious from the mechanical indicator that such was not the case. No serious attempt was made to get exact results by this method as the load could not be determined accurately at the low pressures involved, and also because study on the subject indicates that this method of attack is fallacious for the following reasons:

(a) When the critical condition is approached the film thickness is very small, so small in fact that its dielectric strength is very uncertain. This uncertainty of dielectric strength is not a matter of the oil thickness alone. Mr. J. L. R. Hayden and Mr. W. N. Eddy (Jour. A. I. E. E., Vol. XLI, Feb. 19[illegible]) conclude that "The disruptive break-down of oil under dielectric stress is not due to the voltage exceeding the dielectric strength of oil, as is the case in air, but is due to something being carried into the dielectric field, or being produced in the

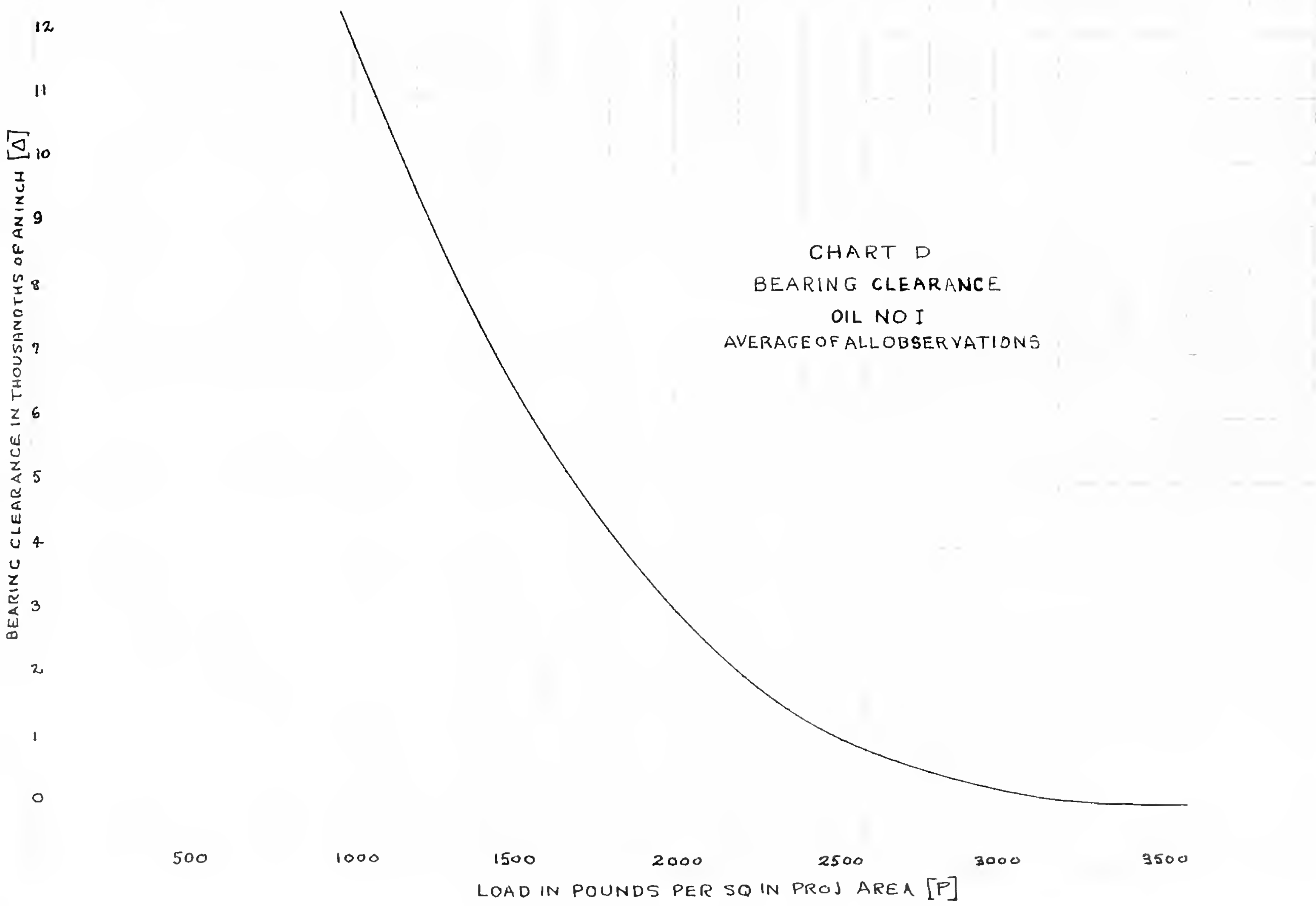


CHART D
BEARING CLEARANCE
OIL NO I
AVERAGE OF ALL OBSERVATIONS
BEARING CLEARANCE IN THOUSANDTHS OF AN INCH [Δ]
0
1
2
3
4
5
6
7
8
9
10
11
12
500
1000
1500
2000
2500
3000
3500
LOAD IN POUNDS PER SQ IN PROJ AREA [P]

dielectric field, which weakens the dielectric strength so as to cause a premature break-down".

(b) Metallic particles which are always present in the oil and which are being constantly circulated in the system would be fatal to this method if they were carried into the bearing. These particles, although invisible to the unaided eye, are still large enough to bridge the film and cause current to flow.

(c) It was demonstrated repeatedly in the trials that when the break-down pressure was approached, contact would occur at points and would evince itself by an unsteadiness of the pendulum. These periods were of short duration, without exception, and the conclusion is that they were caused by small excrescences of the bearing surface that would be smoothed off after a few moments. The false points would often occur when the pressure had reached only a small fraction of the real break-down pressure. For these reasons, it is evident that if the electrical method were depended on, it would give entirely erroneous results.

Determination of Film Thickness.

During the final trial a careful determination was made of the relative position of the two halves of the bearing, by means of the micrometers. The averages of all observations made while using oil No. 1 are shown on Chart L.

In view of the fact that the clearance in the bearing was maintained entirely by the pressure of the oil film, it is thought that the values determined are a fair approximation of the oil film thickness. The method did not admit of very close-measuring of the movements of the bearing blocks near the critical point and consequently the magnitude of the oil film in this region could not be determined with great exactitude.

Conclusions.

The most important fact brought out in this work is the astonishing intensity of pressure that the oil film will stand before being displaced. Even under the most adverse conditions of operation in the experimental work, it is seen that the actual break-down pressures are several hundred times [illegible] the values which have hitherto been accepted as possible. No indication of this has been recorded in the work of previous experiments with cylindrical bearings, so it is thought that this point can be claimed as an original contribution to the knowledge on the subject.

The relationship between break-down pressure, speed and viscosity was not definitely established, but when [illegible] [illegible] acc-

[illegible] to indicate [illegible] to develop its general trend.

The [illegible] break-down [illegible] completely dislodged and the bearing [illegible] journal should not be considered as the critical pressure but [illegible] the point at which metallic contact first makes itself manifest by the unstableness of the pendulum. This point should mark the limit, at least as far as economic loading of the bearing is concerned. It was deemed impractical if not impossible to determine this point exactly with the present apparatus, because the mechanism recording deflections of the pendulum was not delicate enough to indicate these first contacts, so in order to get an unmistakable reference point the actual seizing pressure had to be taken. It should be noted, however, that the points at which unstableness was first noted were in no case far removed from the seizing points.

The results with the oils treated with the oleic acid indicate that the break-down pressures with the oils so treated are less than the straight mineral oils. The experimental results in this direction were meager and it would not be safe to claim that this is invariably true without further confirmation. The inference from this evidence cannot be avoided, that oils having the same absolute viscosity but differing in chemical consistency do not have the same resistance to rupture and hence, from this point of view, differ in lubricating value.

In the original design of the experimental apparatus the necessary devices were developed for determining the essential elements for the complete exposition of the hydrodynamical theory of lubrication and with suitable modifications and refinements in a redesign of the apparatus every premise in the theory could be carried to a successful conclusion. In order that these modifications and refinements can be definitely fixed, they will be dealt with in detail in the following discussion:

Extension of the Experimental Work and Redesign of the Test Apparatus.

Lack of time prevented carrying the investigation to an entirely satisfactory conclusion and, in fact, it is believed that further work would prove abortive with the apparatus as it stands, due to its limitations. It is thought, however, that these could be entirely overcome by a redesign.

The inherent defects of the machine will be pointed out so that if it is considered desirable to continue this line of attack, these could be corrected when the machine is rebuilt.

(a) The entire structure is too flimsy. This is particularly

true of the structural elements of the pendulum. In the original design a maximum pressure of 15,000 pounds was contemplated but as a matter of fact, in the actual runs, loads of 12,000 pounds were not unusual.

(b) The speed control was poor and it was found impossible to operate satisfactorily at speeds lower than 500 R. P. M., as the power of the motor was so feeble at these slow speeds that small increases in the friction was enough to make a large change in the speed and in consequence observations taken at these low speeds were unreliable. This could be readily overcome by installing a motor of ample power and speed, fitted with the proper kind of speed control. A wide range of speed could be obtained by step pulleys such as are fitted to lathes, and very low speeds could be realized by back gears. A hydraulic, friction cone, or some other similar variable speed control could possibly be substituted for the back gears.

(c) The self-aligning feature of the bearing should be carefully considered, and made in a manner to insure the proper functioning of this feature under all conditions of load.

(d) A more accurate value of the actual temperature existing in the oil film should be obtained. This could be accomplished by imbedding several thermo-couples or resistance thermometers in the actual bearing surface.

(e) The bearing back should be cored and arranged with a water cooling service, so that the temperature of the bearing could be controlled. This could be accomplished by using connections of flexible metallic tubing. The effort of the pendulum is so great that any friction involved would not affect its displacement.

(f) Another desirable feature would be a quick releasing device for the loading mechanism so that the load could be quickly removed. This could be done by tapping into the space under the piston and fitting the jack with a small auxiliary piston that could be worked with the fingers very quickly to release the pressure, and such a device would serve also for making fine adjustments of the load.

(g) A cut-out switch could be provided so that when the pendulum is displaced beyond a certain limiting angle the machine will automatically shut itself down.

(h) A multiplying device to indicate small changes or unstableness in the deflection of the pendulum should be installed so that the moment at which the film first begins to rupture can be detected with accuracy.

(i) In order [illegible], and length of bearing [illegible] journal [illegible] could [illegible] in the [illegible] of sleeves that could [illegible] the spindle and [illegible] that they could be secured to a collar on the spindle by means of [illegible]. A [illegible] of [illegible] bearings of various [illegible] could also be provided so that the [illegible] and length of the journal and bearing could be varied at will.

(j) With the apparatus modified as described, a systematic method of testing could be followed, establishing definitely the various factors necessary to the complete solution of the hydrodynamical theory, viz: (1) Film thickness. (2) Temperature and pressure of film and consequently its absolute viscosity. (3) Intensity of pressure on unit area. (4) Rubbing velocity. (5) Diameter and length of journal and bearing.

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Dr. Robert B. Roulston, Associate Professor Thompson and Messrs. Medaugh and Comber, under whom I pursued very interesting and valuable courses, were especially considerate and this acknowledgment would not be complete without recording my appreciation of the interest shown by the entire faculty.

BIOGRAPHICAL SKETCH.

Leonard Noel Linsley was born on the 24th of April, 1885, at Baltimore, Maryland. He received his early education in the public schools of Baltimore and entered the Navy of the United States, in February, 1903. After passing through the various subordinate grades, in due course, he was commissioned a lieutenant commander in 1918.

After completing a preliminary course at the Naval War College in 1914, he was selected by the Navy Department to pursue post graduate studies at the Naval Academy and at Columbia University. In June, 1916, after satisfactorily completing the prescribed courses, he was admitted to the degree of Master of Science in Columbia University.

In the fall of 1919, he entered into residence at the Johns Hopkins University as a graduate student. In the academic years 1919-20, 1920-21, he pursued studies in languages and advanced technical subjects, and in 1921-22, devoted his time to research. In addition to his university work, he conducted night classes for the instruction of officers of the Naval Reserve.

During the Great War he served as a lieutenant commander on the United States Destroyer Stockton, in the capacity of executive officer and navigator and when that vessel was temporarily put out of action, he was assigned to duty by the British Admiralty as an observer on H. M. S. Bruce, a British flotilla leader of the latest type. For a period of nearly two years he saw constant service in the war zone and took part in several engagements with the enemy warships.

In the spring of 1919 he was executive officer of the destroyer which accompanied the U. S. Naval Aeroplane, NC 4, in its successful flight across the Atlantic.

During his service in the Navy he has twice circumnavigated the globe; has served on every type of vessel and in every clime; has seen active service in the Near East, China, the Phillipines, Cuba, Haiti, Nicaragua and Mexico.